Design and Development of Axial Feed Using Recirculating Ball Screw for 6-Axis CNC Gear Hobbing Machine

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Abstract - *The present paper highlights gear forming machines and in particular to those machines which forms gears by the hobbing process, utilizing tools having helically arranged stock removing surfaces.*

Hobbing is primarily used for producing spur and helical gears; however, other products, such as worm wheels, sprockets, and splined shafts may also be produced by the hobbing process. This process is inexpensive and accurate compared to other gear forming process. Present paper focuses on voluminous study of Recirculating ball screw over the conventional lead screw systems. Recirculating ball screw system features high precision and accuracy level compared to lead screw system. Backlash error extermination and high precision finished product has been achieved after utilization of Recirculating ball screw.

We have designed an Axial Feed drive system for a LEIBHERR CNC L400 gear hobbing machine as well as we have carried out various design validation processes to provide high results. On the basis of the feed system, stroke length, mounting space, precision and life we have selected a suitable Recirculating ball screw system from the catalogue.

***Key Words*: Recirculating Ball Screw System[1], Backlash Error[2], Hobbing Machine[3], Shaft Length[4], Bearing Housing[5], Thrust Bearing[6]**

1. INTRODUCTION

The hobbing process, a cylindrical-shaped rotating tool having helically arranged material removing surfaces is brought into contact with a rotating workpiece, generally a gear blank. In spur gear hobbing, the tool and workpiece rotate in a timed relationship as though the workpiece were a gear rotating in mesh with is a worm gear represented by the hobbing tool. When helical gears are hobbed, a supplemental rate of motion is applied to the workpiece rotation, either advancing or retarding the fundamental timing relative to hobbing tool rotational rate, in order to develop the appropriate helix angle across the face width of the gear teeth being machined.

Generally, long cylindrical hob is used, which is shifted along its axis either periodically in small augmentations or continuously during intervals between cutting operations in order to allow all regions along its length to cut and thus spread the tool wear over the entire tool to increase the tool life. The purpose of axially shifting the tool is to avoid excessive wear on hob cutter. The hob slide is ordinarily mounted upon an angularly adjustable hob head, which swivels about an axis perpendicular to and intersecting the hob's central axis.

A ball screw uses recirculating ball to minimize friction and maximize efficiency while a lead screw depends on low coefficients of friction between sliding surfaces. A lead screw therefore typically cannot achieve the efficiency of a ball screw (≈95%). The study of wear and friction leads one to conclude that sliding friction is inherently less predictable than power transmission utilizing recirculating ball technology. Thus, there is a fundamental difference in application of a ball screw and a lead screw due to the ability to predict performance and life.

With all the advantages of a ball screw (load capacity, rigidity, efficiency) there is a price to be paid. Although their performance to cost ratio is very high when compared to other means of translating linear motion, a ball screw design is more complex, requiring hardened precision bearing surfaces and a ball recirculation mechanism. On the other hand, a lead screw is very compact, offers great design flexibility, and does not produce sound, generally corrosion resistant. They are very capable in many applications, but they do have their limitations as well.

The present article focuses on design and development of axial feed for 6-axis gear hobbing machine using recirculating ball screw system. The hobbing machine comprising a column, an axial slide mounted to the column so as to be vertical travel with respect to the column, a hob head mounted to the hob holder so as to be horizontal travel with respect to the axial slide, parallel to and fro with respect to bed, an arbor mounted operatively to the head and rotatable about a generally horizontal axis, and a motor for rotating the arbor about the generally horizontal axis. Later on, the ball screw was selected on various parameters like operating conditions, operating space available, feed rate, accuracy, stroke length and life, etc. Hobbing machines are available as Non-CNC Type and CNC type hobbing machine.

1.1 Types of Feed in CNC Gear Hobbing Machine

The feed types in a CNC type Gear Hobbing Machines are:

- Radial Feed for Bed
- Axial Feed for axial slide
- Tangential feed for hob
- Clockwise/ Anti-clockwise Feed for worktable
- Rotational feed for hob cutter along its axis
- Swiveling axis for hob head

So, this study will include the modeling of an axial feed drive in an Industrial CNC Gear Hobbing Machine (as shown in Fig. 1 to 3).

1.2 Design Process for an Axial Feed

The Modelling of axial feed in machine is done in two phases:

1. Selection of Ball Screw.
2. Selection of Bearings.
3. Selection of Bearing Housing.
4. Lubrication and contamination protection

The Detailed Design and Modelling procedure along with suitable calculations are as follows.

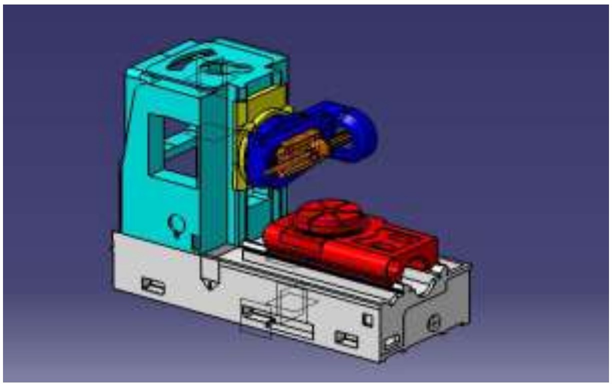


Fig -1: Assembly of 6-Axis CNC Gear Hobbing Machine

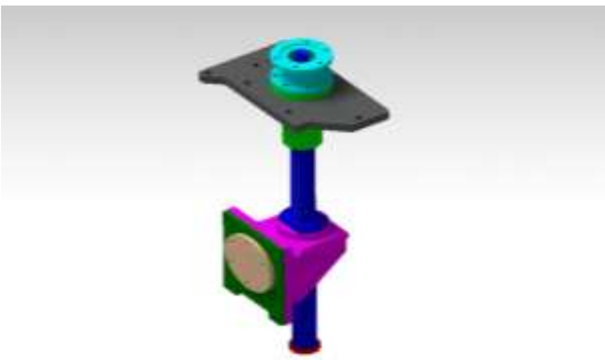


Fig -2: Assembly of Ball Screw along with Mounting Bracket and Bearing Housing

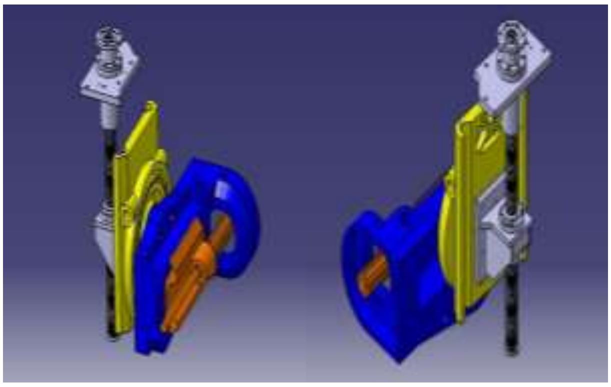


Fig -3: Orientation & Load Conditions over Ball Screw

2. CALCULATIONS OF VARIOUS DESIGN PARTS USED IN THE AXIAL FEED DRIVE:-

2.1. Phase 1 - Selection of Recirculating Precision Ball Screw:-

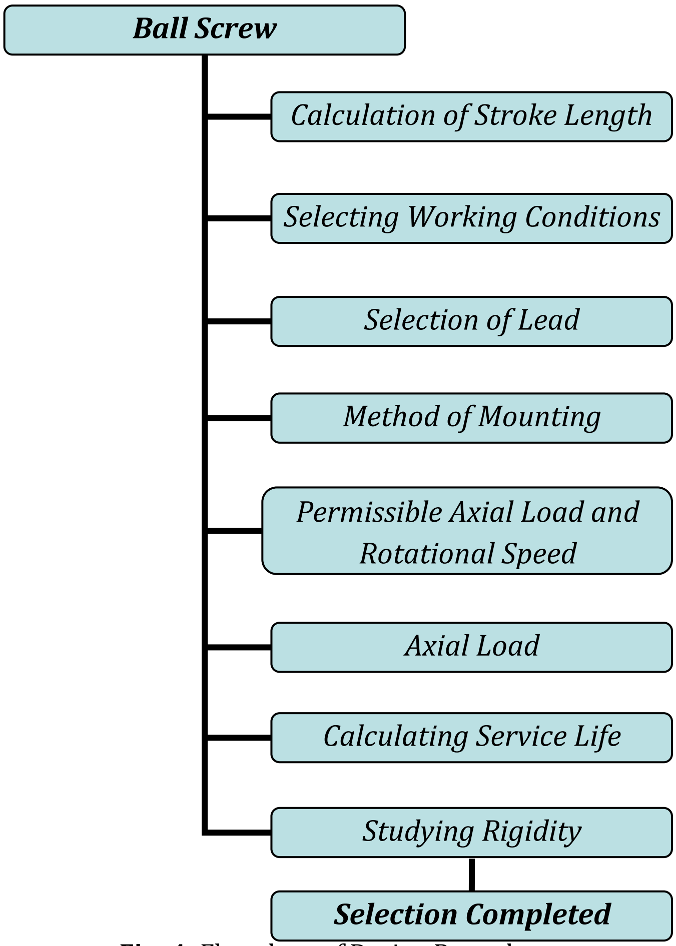


Fig -4: Flowchart of Design Procedure

2.1.1 Calculation of Stroke Length:-

For calculating of stroke length, the layout of screw shaft (as shown in Fig. -5) should be analyzed.
Maximum Stroke of Axial Slide: 340 mm

Minimum Stroke of Axial Slide: 40 mm

Total Effective Stroke of the Axial Slide: 340 – 40 = 300 mm

Nut Length: 181 + 19 = 200 mm

Clearance Distance: 30 mm

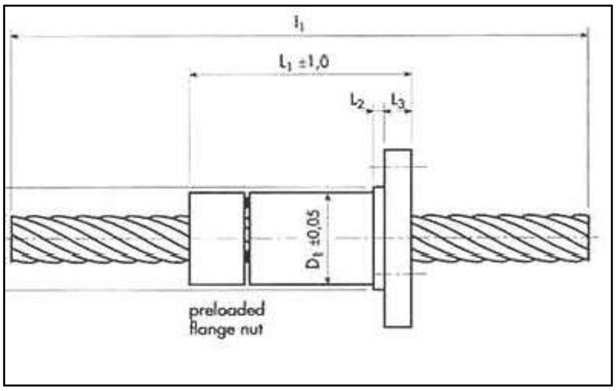


Fig -5: Layout of Shaft Design

Total length of Ball Screw: 300 + 200 + 30 = 530 mm

Take total Safe length: 550 mm

Length of Total shaft: 550 mm

2.1.2 Selecting Working Conditions:-

Transfer orientation: Vertical

Transferred Mass, m = 800 Kg

Guide Method: Sliding

Coefficient of Friction, μ = 0.15

Guide Surface Resistance, f = 20 N

Stroke Length, L_s= 300 mm

Speed, V_{max} = 15 mm/sec

Number of Reciprocations, n = 1

Reduction Gear Ratio, A = 20

Speed of AC Servo-Motor, N_{motor} = 2000 rpm

2.1.3 Selection of Lead, P_h:-

$P_h = (V * 1000 * 60) / N_{i/p}$

P_h = 9 mm

$P_h \approx 10$ mm

2.1.4 Method of Mounting:-

Fixed- Free: $\eta_1 = 0.25$ $\eta_2 = 1.3$

Fixed- Supported: $\eta_1 = 2$ $\eta_2 = 10$

Fixed- Fixed: $\eta_1 = 4$ $\eta_2 = 20$

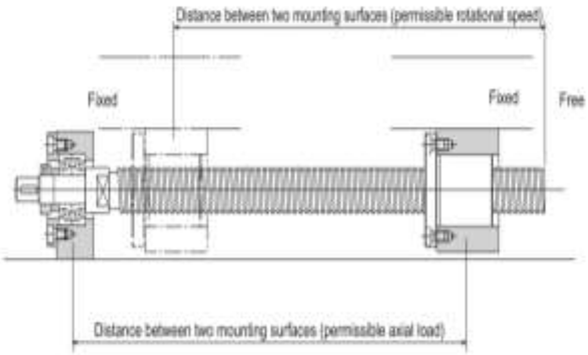


Fig -6: Screw Shaft Mounting Method (Vertical)

*Due to space restrictions and according to user application, the method selected is Fixed-Free type in vertical orientation (as shown in Fig. -6).

2.1.5 Permissible Axial Load & Rotational Speed:-

A) Buckling Load:-

$P_1 = (\eta_2 * d_1^4 * 10^4) / L^2$

$P_1 = (1.3 * (57.856)^4 * 10^4) / 800^2$

P_1 = 227.592 KN

η_2: Factor according to Mounting Method

d_1: Screw Shaft Thread Minor Diameter

L: Distance between Two Mounting Surfaces

B) Tensile Compressive Load:-

$P_2 = 116 * d_1^2$

P_2 = 388.288 KN

C) Maximum Rotational Speed:-

$N_{max} = (V_{max} * 60 * 10^3) / P_h$

$N_{max} = (0.015 * 60 * 1000) / 10$

N_{max} = 90 Revolution per Minute

D) Permissible Rotational Speed Determined by Dangerous Speed of Shaft:-

$N_1 = (\lambda_2 * d_1 * 10^7) / L^2$

$N_1 = 3.4 * 57.856 * 10^7 / 800^2$

$N_1 = 3073.6$ Revolution per Minute

λ_2: Factor according to Mounting Method

Fixed- Free : 3.4

Fixed- Supported : 15.1

Fixed- Fixed : 21.9

E) Permissible Rotational Speed Determined by DN Value:-

D: Ball Center to Center Diameter

$N_2 = 70000 / D$

$N_2 = 70000 / 65$

$N_2 = 1076.92$ Revolution per Minute

Thus, with a ball screw having screw shaft diameter of 63 mm and a lead of 10 mm, the maximum rotational speed is less than the Dangerous Speed and the DN value.

2.1.6 Axial Load:-

A) Acceleration:-

$\alpha = V_{max} / t_1$

$\alpha = 0.015 / 0.2$

$\alpha = 0.075\ m/s^2$

B) Forces:-

During Upward Acceleration:

$Fa_1 = mg + f + m\alpha$

$Fa_1 = 7928$ N

During Upward Uniform Motion:

$Fa_2 = mg + f$

$Fa_2 = 7868$ N

During Upward Deceleration:

$Fa_3 = mg + f - m\alpha$

$Fa_3 = 7808$ N

During Downward Acceleration:

$Fa_4 = mg - f - m\alpha$

$Fa_4 = 7768$ N

During Downward Uniform Motion:

$Fa_5 = mg - f$

$Fa_5 = 7828$ N

During Downward Deceleration:

$Fa_6 = mg - f + m\alpha$

$Fa_6 = 7888$ N

Thus, the maximum axial load applied on the Ball Screw is as follows: $Fa_{max} = Fa_1 = 7928$ N

C) Travel Distance:-

Maximum speed, $V_{max} = 0.015$ m/s

Acceleration time, $t_1 = 0.2$ second

Deceleration time, $t_3 = 0.2$ second

Travel distance during acceleration:-

$l_{1,4} = (V_{max} * t_1 * 10^3) / 2$

$l_{1,4} = (0.015 * 0.2 * 1000) / 2$

$l_{1,4} = 1.5$ mm

Travel distance during uniform motion:-

$l_{2,5} = l_s - (V_{max} * t_1 + V_{max} * t_3) * 1000/2$

$l_{2,5} = 300 - \{(0.015 * 0.2 + 0.015 * 0.2) * 1000/2\}$

$l_{2,5} = 300 - 3$

$l_{2,5} = 297$ mm

Travel distance during deceleration:-

$l_{3,6} = (V_{max} * t_3 * 10^3) / 2$

$l_{3,6} = (0.015 * 0.2 * 1000) / 2$

$l_{3,6} = 1.5$ mm

D) Average Axial Load:-

$F_m = \{(Fa_1^3 * l_1 + Fa_2^3 * l_2 + Fa_3^3 * l_3 + Fa_4^3 * l_4 + Fa_5^3 * l_5 + Fa_6^3 * l_6) / (l_1 + l_2 + l_3 + l_4 + l_5 + l_6)\}^{1/3}$

$F_m = 7848.06$ N

E) Allowable Axial Load:-

Assuming that an impact load is applied during an acceleration and a deceleration, set the static safety factor (f_s) at 3

$Fa_{max} = C_0a / f_s$

$Fa_{max} = 275617 / 3$

$Fa_{max} = 91.872$ KN

The obtained permissible axial load is greater than the maximum axial load of 7928 N, and therefore, there will be no problem with this model.

2.1.7 Calculating Service Life:-

A) Nominal Life:-

Dynamic load rating, Ca= 76418 N

Load factor, f_w= 1.2

Average load, F_m = 7848.06 N

Nominal life, L' (rev) =?

$L' = (Ca / f_w * F_m)^3 * 10^6$

$L' = (76418 / 1.2 * 7848.06)^3 * 10^6$

$L' = 5.34 * 10^8$ revolutions

B) Average Revolutions per Minute:-

Number of reciprocations per minute, n = 1 min^{-1}

Stroke, L_s= 300 mm

Lead, P_h= 10 mm

$N_m = (2 * n * L_s) / Ph$

$N_m = (2 * 1 * 300) / 10$

$N_m = 60\ min^{-1}$

C) Calculating the Service Life Time:-

Nominal life, $L' = 5.34 * 10^8$ rev

Average revolutions per minute, $N_m = 60\ min^{-1}$

$L_h = L' / (60 * N_m)$

$L_h = (5.34 * 10^8) / (60 * 60)$

L_h = 148333.3 hours

D) Calculating the Service Life in Travel Distance:-

Nominal life, $L' = 5.34 * 10^8$ revolutions

Lead, P_h= 10 mm

$L_s = L * P_h * 10^{-6}$

$L_s = 5.34 * 10^8 * 10 * 10^{-6}$

L_s = 5340 Km

2.1.8 Studying Rigidity:-

To increase the positioning accuracy and to reduce the displacement caused by cutting force, rigidity of selected ball screw is studied.

For Fixed- Free Configuration:

$K_s = (A * E) / 1000 * L$

$K_s = (3347.32 * 2.06 * 10^5)/ 1000 * 800$

K_s = 861.934 N/μm

A: Screw shaft cross-sectional area = $(\pi / 4) * d_1^2$, mm^2

E: Young's Modulus, N/mm^2

Based on above calculations, suitable Precision Ball Screw was selected from manufacturer's catalogue, Double Rotating Nut Precision Ball Screw with Central Flange (as shown in Fig. – 7).

Korta Group – Precision Ball Screw – Double Nut with Central Flange

Model No: EDBS-6310

Fig -7: Double Rotating Nut Precision Ball Screw with Central Flange

2.2 Phase 2 - Selection of Bearings:-

Bearings:

1) **6208-2Z**
2) **6209-2Z**
3) **81108 TN**
4) **81109 TN**

Table -1: Specification of Bearing

Parameter / Bearing No.	6208-2Z	6209-2Z	81108	81109
d (mm)	40	45	40	45
D (mm)	80	85	60	65
B (mm)	18	19	13	14
C (KN)	32.5	35.1	43	45
C_o (KN)	19	21.6	137	153

2.2.1 Deep Groove Ball Bearing:-

$F_r = 650$ N
$R_A + R_B = F_r$

$\sum M_A = 0$
$(R_B * 174) - (F_r * 974) = 0$
$R_B = 3638.5$ N
$R_A = -2988.5$ N

A) For $F_{r1} = 3000$ N

Bearing: 6208-2Z

Specification:
d = 40 mm
D = 80 mm
B = 18 mm
C = 32.5 KN
$C_o = 19$ KN

$F_a <<< F_r$
Axial Load Neglected
$P=F_r=3000$ N

$L_{10}= (C / P)^3$

$L_{10} = 1271.412$ Million Revolution

$L_{10h} = (L_{10} * 10^6) / (60 * N)$

$L_{10h} = 211902$ Hours
$L_{10h} = 24$ Years

B) For $F_{r2} = 3650$ N

Bearing: 6209-2Z

Specification:
d = 45 mm
D = 85 mm
B = 19 mm
C = 35.1 KN
$C_o = 21.6$ KN

$F_a <<< F_r$
Axial Load Neglected
$P=F_r=3650$ N

$L_{10} = (C / P)^3$

$L_{10} = 889.288$ Million Revolution

$L_{10h} = (L_{10} * 10^6) / (60 * N)$

$L_{10h} = 14821.4.7$ Hours
$L_{10h} = 16.91$ Years

Single Row Deep Groove Ball Bearing is shown in Fig. -8.

2.2.2 Cylindrical Roller Thrust Bearing:-

For $F_a = 5500$ N

A) Bearing: 81108 TN

Specification:
d = 40 mm
D = 60 mm
H = 13 mm
C = 43 KN
$C_o = 137$ KN

$P= F_a=5500$ N

$L_{10} = (C / P)^{10/3}$

$L_{10} = 948.640$ Million Revolution

$L_{10h} = (L_{10} * 10^6) / (60 * N)$

$L_{10h} = 158076.74$ Hours
$L_{10h} = 18.045$ Years

B) Bearing: 81109 TN

Specification:
d = 45 mm
D = 65 mm
H = 14 mm
C = 45 KN
$C_o = 153$ KN

$P = F_a = 5500$ N

$L_{10} = (C / P)^{10/3}$

$L_{10} = 1103.653$ Million Revolution

$L_{10h} = (L_{10} * 10^6) / (60 * N)$

$L_{10h} = 183942.25$ Hours
$L_{10h} = 20.99$ Years

Cylindrical Roller Thrust Bearing is shown in Fig. -9.

Table -2: Final Bearing Selection

Sr. No.	Ball Bearing	Life	Sr. No.	Roller Bearing	Life
1)	6208-2Z	24 Years	**1)**	81108 TN	18 Years
2)	6209-2Z	17 Years	**2)**	81109 TN	21 Years

Fig -8: Single Row Deep Groove Ball Bearing

Fig -9: Cylindrical Roller Thrust Bearing

2.3 Phase 3 - Selection of Bearing Housing:-

- Designing of bearing housing is according to shaft length and selected bearings.
- Basic need for the housing is to convert the rotary motion of shaft into linear motion.
- Alongside, the shaft needs to be rigid enough to sustain applied loads from various parts and processes, which enhances elimination of backlash error.

Further design iterations were done and design was finalized (as shown in Fig. -10).

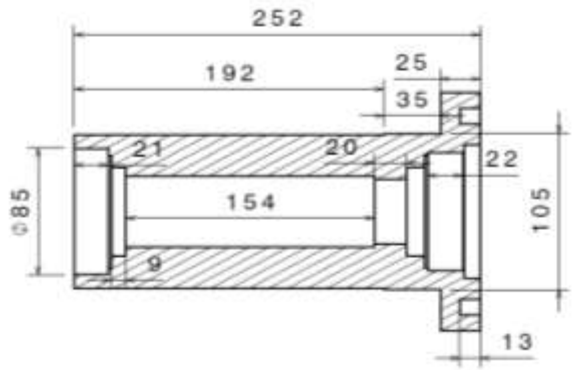


Fig -10: Bearing Housing Draft

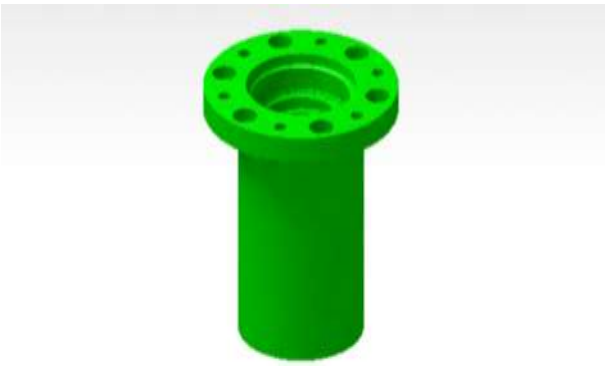


Fig -11: Bearing Housing

Bearing housing (as shown in Fig. -11) is designed according to outside diameter of selected bearings along with spacers and preload caps.
Further, the Ball screw shaft diameter was reduced according to inner diameter of bearings, considering shaft rigidity, load carrying capacity and precision movement of ball screw.

2.4 Phase 4-Lubrication and Contamination Protection:-

Lubrication:

To maximize the performance of ball screw, it is necessary to select proper lubricant and lubrication method. With this, right amount of lubricant is provided to the raceway of ball screw shaft; this allows an oil film to be constantly formed between the balls and raceway, improves lubricity and extends the lubrication maintenance interval.

Contamination Protection:

If foreign material enters the interior of ball screw, abnormal levels of abrasion and ball clogging is likely to occur. This can shorten overall lifespan of ball screw. As such, foreign material needs to be prevented from entering. So, it is important to choose an effective contamination protection product that suits the usage conditions.

Hence, the selected ball screw is protected with labyrinth type of seal that is designed to protect the foreign particles from entering the screw shaft, also it consists of internal grease lubrication which helps is eradication of heat generation, when in operation.

3. ANALYSIS

Once design was finalized we carried out suitable analysis of parts to accurately define our design. As the shaft was been outsourced, we had to carry out modal and static analysis to check the maximum frequency, maximum stress generated in the ball screw shaft.

3.1 Modal Analysis

Modal analysis is a simple way to calculate the natural frequencies of your system so as to know which frequencies can be destructive and dangerous for the part. Modal analysis calculates the natural frequencies of the system alone. Modal is the simplest analysis and the only thing it does is telling you what are the "resonance frequencies" of your geometry. Values shown in ANSYS specify the maximum frequency the shaft can sustain before failure.

We have used gear drive system to transmit the power from the motor to the driven shaft. The motor runs at 2000 rpm and transmits a rotation of 100 rpm to the ball screw shaft. The shaft assembly is been connected with the gearbox. Hence the entire shaft assembly rotates at 100 rpm. The solution provided by the software ANSYS 15.0 will be in Hertz. It provides us a range of frequency from maximum to minimum values. So converting Revolution per Minute to Hertz, we get 100 rpm = 1.67Hz. With maximum of 2000 rpm, the design must be safe for a frequency of 33.33 Hz.

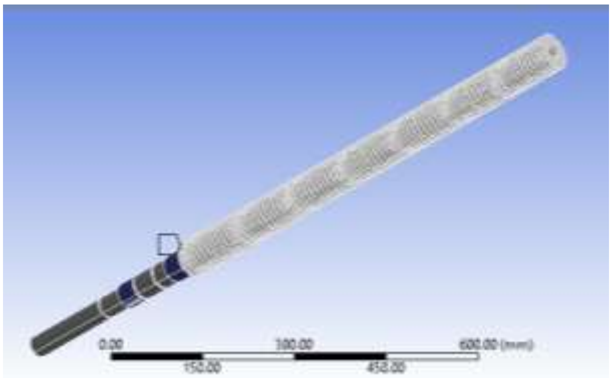


Fig -12: Boundary Conditions

Here, the highlighted portion is the fixed support in the ball screw model (as shown in Fig. -12). As this part is been fixed into the assembly with the bearing housing, we carried out analysis for 6 different modes of frequency (as shown in Fig. -13 to 18) over the ball screw shaft design.

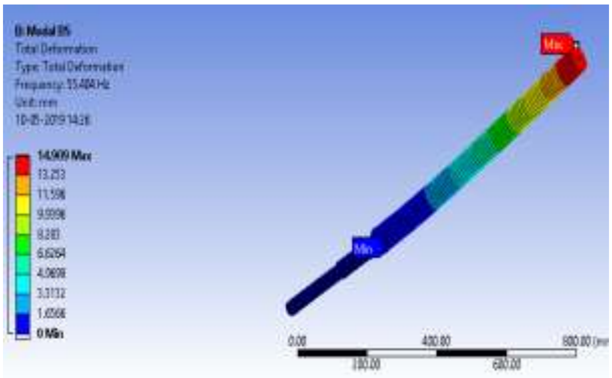


Fig -13: Mode 1

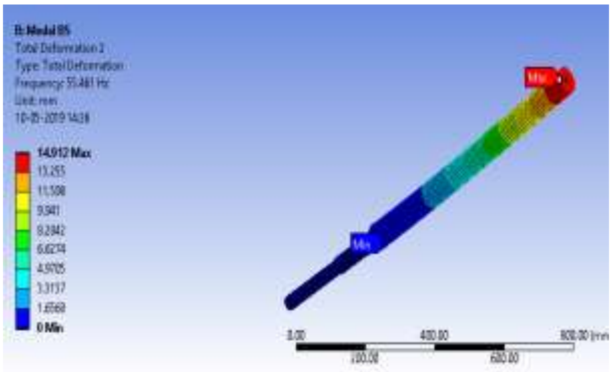


Fig -14: Mode 2

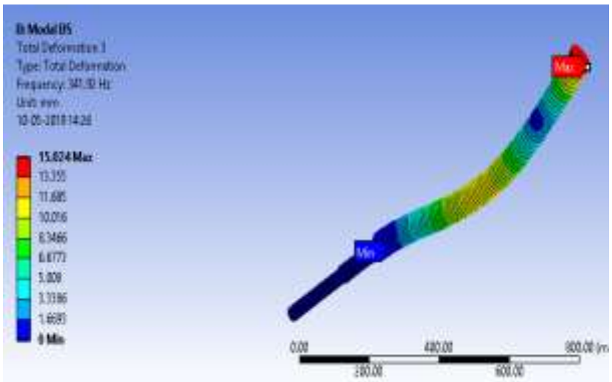


Fig -15: Mode 3

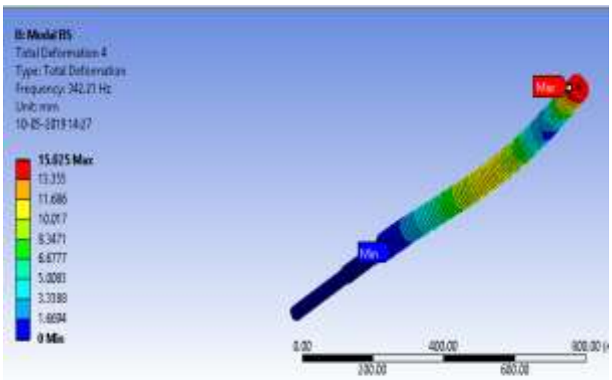


Fig -16: Mode 4

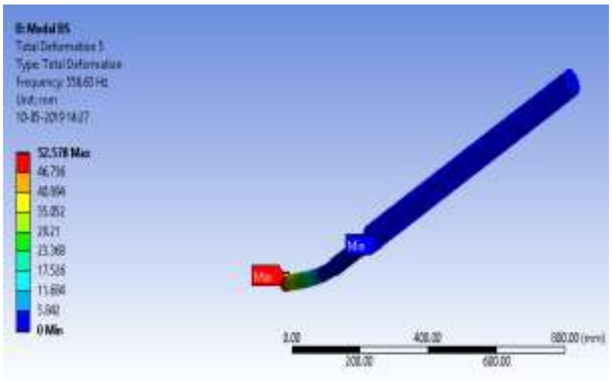


Fig -17: Mode 5

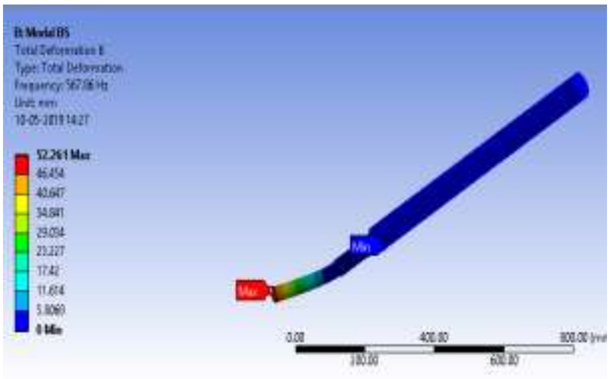


Fig -18: Mode 6

Tabular Data

	Mode	Frequency [Hz]
1	1.	55.404
2	2.	55.461
3	3.	341.92
4	4.	342.21
5	5.	556.63
6	6.	567.86

Fig -19: Frequency Table

Converting RPM to HZ,

Minimum: 100 rpm = 1.67 Hz.

Maximum: 2000 rpm = 33.33 Hz.

So, the calculated frequency value (as shown in Fig. -19) does not coincide with natural frequency values obtain from ANSYS 15.0. This concludes that resonance will not occur for given range of speeds.

3.2 Static Analysis

Structural analysis is mainly concerned with finding out the behavior of a physical structure when subjected to force. A static load is one which varies very slowly. A dynamic load is one which changes with time fairly quickly in comparison to the structure's natural frequency. If it changes slowly, the structure's response may be determined with static analysis, but if it varies quickly (relative to the structure's ability to respond), the response must be determined with a dynamic analysis.

After modal analysis, we decided to carry out the static analysis of the shaft assembly to determine its FOS and Service Life. First, we calculated the forces acting over the screw shaft. Later we generated a suitable mesh (Shown in Fig. -20) over the geometry. Further, loading conditions are applied on the screw shaft according to the acting forces. Material used for the ball screw shaft was AISI 6150.

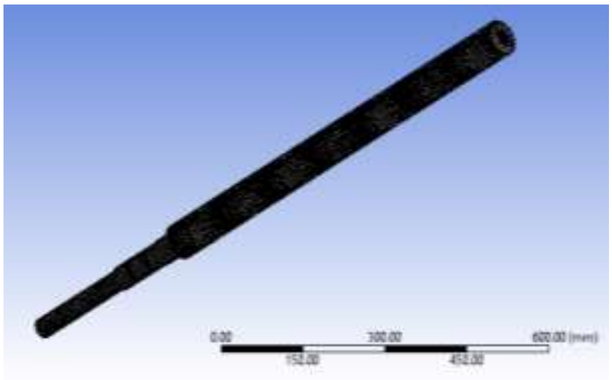


Fig -20: Meshed Geometry

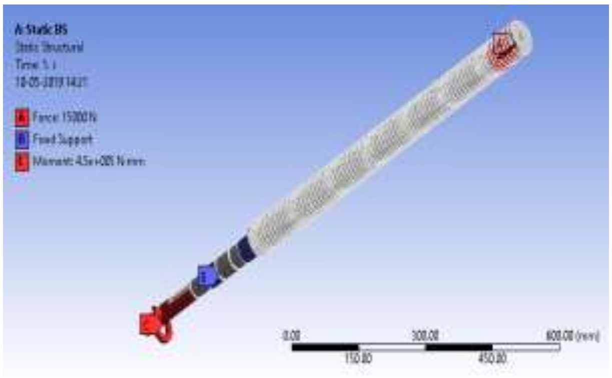


Fig -21: Loading Conditions

1. Force :- 15000 N (Axial)
2. Moment :- 450 Nm
3. Fixed Support

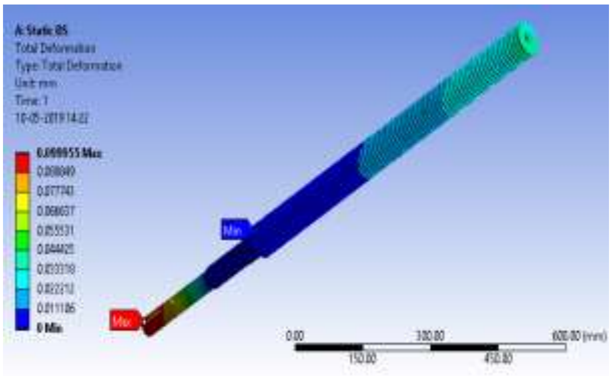
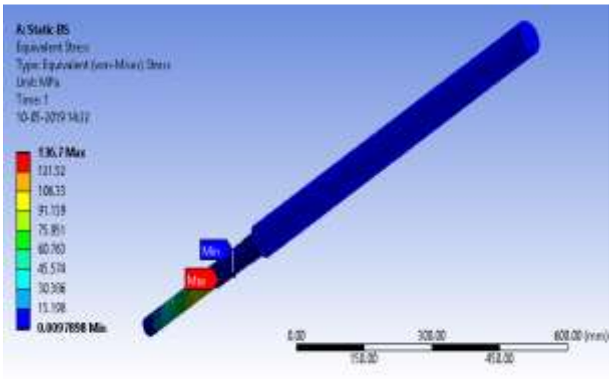


Fig -22: Total Deformation



. **Fig -23**: Equivalent (von-Mises) Stress

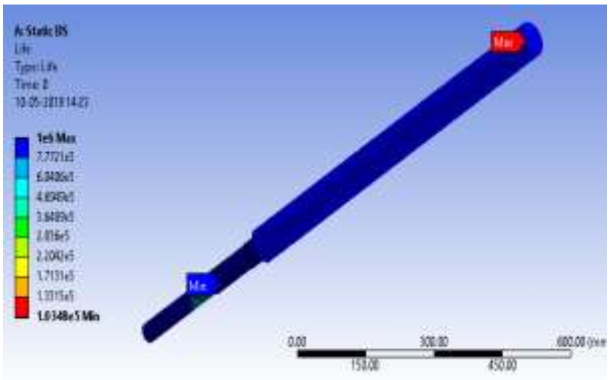


Fig -24: Fatigue Life

3.3 Results:

Using the software ANSYS 15.0, total deformation, equivalent stress (von-mises) and fatigue life were analyzed (as shown in Fig. -22 to 24).

Hence following results were generated from the above analysis:

Maximum Stress: 136.7 MPa

Maximum Deformation: 0.099 mm

Minimum Life: 1.034 X 10^5 cycles

FOS = Yield Strength / Maximum Stress

FOS = 415 / 136.7

FOS = 3.04

Hence our design is considered to be safe as the generated FOS is 3.04 > 2.0

4. CONCLUSIONS

We have designed an Axial Feed System using Recirculating Ball Screw for 6- axis CNC Gear Hobbing Machine. We carried out various iterations on selection of Ball Screw and Bearings from manufacturer's catalogue according to application. We also carried out iterations over design of bearing housing to hold the ball screw good.

Using Ball screw systems, we successfully reduced backlash error which was introduced in lead screw systems. Error reduced from 0.8mm to 0-10 microns.

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*For reference [4] to [10], website (with given URL) were considered to obtain the standard catalogue of required product like screw shaft, deep groove ball bearing & cylindrical thrust bearing that were provided by company according to the industrial standards. Further, product specifications were used for calculation, cad design, modeling and analysis of the part for the safe selection of product for required operation in designed system.

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